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TECHNICAL NOTE

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THE DESIGN OF VARIOUS TYPES OF AIR BEARINGS FOR SIMULATING FRICTIONLESS ENVIRONMENTS

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SUMMARY

Several types of air bearings are discussed and analyzed which can effectively simulate a frictionless environment for testing space vehicle instruments and control systems. A spherical dual-flow bearing, designed for a load of 15 pounds and employing an input pressure of about 4.5 psi, has been operated with an effective coefficient of friction of only 0.00000406. However, unless all externally applied torques as well as the center of gravity of the test fixture are exactly at the bearing's center of rotation, it will precess. This is overcome in a cylindrical dual-flow bearing designed on the same principle. A spherical mono-flow bearing and a flat plate mono-flow bearing, which will bear loads of 700 and 400 pounds respectively with low input pressures, are also discussed. Plots of theoretical and actual performance are given, and fabrication techniques are described.

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INTRODUCTION

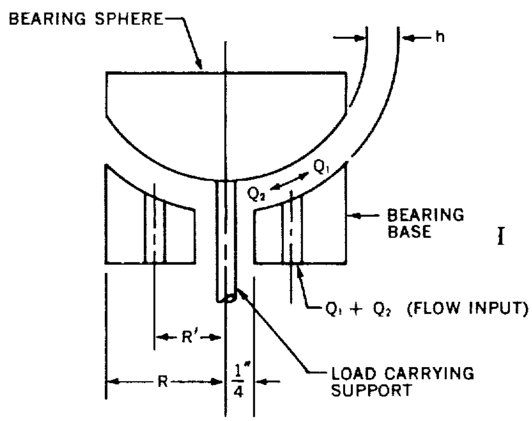
Precision instruments and control systems designed for space vehicles and stabilized orbiting spacecraft must be tested and evaluated in a simulated space environment. In this connection it is necessary to find a means of simulating a frictionless environment. The application of a properly designed air bearing is a highly feasible approach toward creating this environment because the frictional effects of an air bearing can be considered negligible.

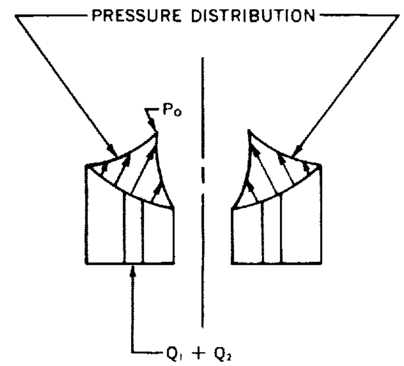
This report describes the design and applications of several types of flat and spherical air bearings. The major discussion is devoted to a dual-flow spherical air bearing: a spherical mono-flow and a flat mono-flow bearing are treated in less detail.

SPHERICAL DUAL-FLOW AIR BEARING

The spherical dual-flow air bearing is so named because air flows into the bearing through capillary tube orifices flush with its inner socket surface between the two exhausts or outlets. One of the outlets is at the outer circumference of the bearing and the other is at the center where the load-supporting rod joins the bearing sphere (Figure 1).

The application of this type of air bearing was introduced when accurate evaluation of various satellite spin reduction mechanisms required an essentially frictionless environment. This environment can be approached with the air bearing, but ideally the entire system should be operated in a vacuum, since the drag effect of air on the device being tested tends to obviate the frictionless condition.





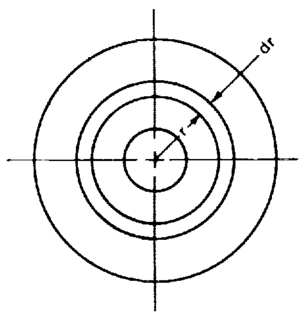


Figure 1 — Configuration and pressure distribution of the spherical dual-flow bearing

An interesting feature of the spherical dual-flow bearing is the small air pressure and flow required to support a given load. The actual design calculations given below, which demonstrate the feasibility of such a system, were based on locating the capillaries so that the flow toward the bearing center equals the flow outward. Figure 1 shows the flow conditions and pressure distribution on which the design was based.

Determination of the Radii

The flow of air through the slot (Figure 1) between the bearing sphere and base is given by the familiar formula*

$$Q = \frac{\Delta P \, b h^3}{12 \mu \ell} \tag{1}$$

where, in this case,

ΔP = the overall pressure differential causing the flow (psi),

$b = 2\pi r$ = the "width" of the circular slot (inches),

$h = h_0$ = the height of the slot (inches),

*Fuller, D. D., "Theory and Practice of Lubrication for Engineers," New York: Wiley, 1956.

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ℓ = dr = the elemental length of the slot (inches),

μ = the viscosity of the air (reyns*).

Note that ΔP is the same in both directions, being the drop from the initial pressure to the outside pressure, and h is a geometrical constant of the system. The capillaries are equally spaced around the bearing center at a distance R'; hence the flows may be considered radially symmetrical. We desire equal flows toward the center and the outside of the bearing; thus

$$Q_1 = \frac{\Delta P\, b_1 h^3}{12\mu\ell_1} = \frac{\Delta P\, b_2 h^3}{12\mu\ell_2} = Q_2 ,$$

from which

$$\frac{dr_1}{r_1} = \frac{dr_2}{r_2} . \tag{2}$$

Integrating Equation 2 from the capillary orifices to the outlets (Figure 1), we have

$$\int_{R'}^{R} \frac{dr_1}{r_1} = \int_{\frac{1}{4}}^{R'} \frac{dr_2}{r_2}$$

$$R' = \frac{\sqrt{R}}{2} . \tag{3}$$

Thus for R = 2 inches (the design value), R' = 0.70 inch.

Determination of the Pressure Distribution

For flow from the orifices to the outside of the bearing, the pressure decreases with increasing radius; thus, from Equation 1,

$$dp = \frac{-12Q\mu\, dr}{2\pi r h^3} . \tag{4}$$

Integrating this equation yields

$$p_1 = -k \ln r + C_1 , \tag{5}$$

*1 reyn = $1 \frac{\text{lb-sec}}{\text{in}^2}$

where

$$k = \frac{6Q\mu}{\pi h^3} .$$

The constant C_1 is determined from the boundary conditions: when $r = R$, then $p_1 = 0$ and

$$C_1 = \frac{6Q\mu}{\pi h^3} \ln R = k \ln R .$$

Equation 5 then becomes

$$p_1 = k \ln \frac{R}{r} . \tag{6}$$

For inward flow from the orifices to the center outlet, a similar analysis shows that

$$p_2 = k \ln r + C_2 . \tag{7}$$

Applying the boundary conditions $r = 1/4$ and $p_2 = 0$ then gives

$$C_2 = -k \ln \frac{1}{4}$$

and thus

$$p_2 = k \ln \frac{r}{1/4} = k \ln 4r . \tag{8}$$

Total Flow and Peak Pressure

We now desire to calculate the peak pressure P_0 required at the input orifices to support the load on the bearing sphere. The design load is, in this instance, $W = 15$ pounds. Let $A = 2\pi r dr$ represent the flat projection of the bearing sphere surface area; then, since $dW = p\, dA$,

$$W = \int_{R'}^{R} p_1 \, 2\pi r \, dr + \int_{\frac{1}{4}}^{R'} p_2 \, 2\pi r \, dr ,$$

where p_1 and p_2 are the pressure distributions as given by Equations 6 and 8.

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Thus

$$W = 2\pi \int_{R'}^{R} \frac{6Q\mu}{\pi h^3} \ln \frac{R}{r} \, r\,dr + 2\pi \int_{\frac{1}{4}}^{R'} \frac{6Q\mu}{\pi h^3} (\ln 4r) \, r\,dr \; . \tag{9}$$

Next an expression for Q in terms of P_0 must be found. From Equation 6, since $P_1 = P_0$ when $r = R'$,

$$Q = \frac{P_0 \pi h^3}{6\mu \ln \frac{R}{R'}} \; . \tag{10}$$

The use of this expression reduces Equation 9 to

$$W = \frac{2\pi P_0}{\ln \frac{R}{R'}} \left(\int_{R'}^{R} r \ln \frac{R}{r} \, dr + \int_{\frac{1}{4}}^{R'} r \ln 4r \, dr \right) .$$

Integrating, solving for P_0, and substituting the numerical values of W, R, and R' then gives

$$P_0 = 2.11 \text{ psi.}$$

The flow Q can be calculated from Equation 10; with $P_0 = 2.11$ psi, $\mu = 2.6 \times 10^{-9}$ reyn (at 70°F), $R = 2$ inches, and $R' = 0.70$ inch, the result is

$$Q = 4.04 \times 10^{8} \, h^3 \, \frac{\text{in}^3}{\text{sec}} \; . \tag{11}$$

The purpose of using capillary tubes is to limit the flow of air when the sphere is not seated in the bearing. The pressure drop across these capillaries can be determined from the familiar flow equation:

$$2Q = Q_{tot} = \frac{\Delta P_c \, \pi R_c^4 N}{8\mu \ell_c} \; , \tag{12}$$

where, in the present case, ΔP_c is the pressure differential, $R_c = 0.004$ inch and

ℓ_c = 0.281 inch are the radius and length of the capillary, and $N = 6$ is the number of capillaries. The result is

$$\Delta P_c = 1.21 \, Q_{tot} .$$

Note that ΔP_c is a function of h. Figure 2 shows both P_c and h plotted against Q_{tot}.

The total pressure differential required across the entire bearing (from entrance to exit) is, of course,

$$\Delta P_{tot} = P_0 + \Delta P_c .$$

The constant k in Equations 6 and 8 can now be evaluated by substituting the values of Q (from Equation 11) and h. For the particular bearing under discussion, $h = h_0$ = 0.001 inch; this results in the pressure distributions

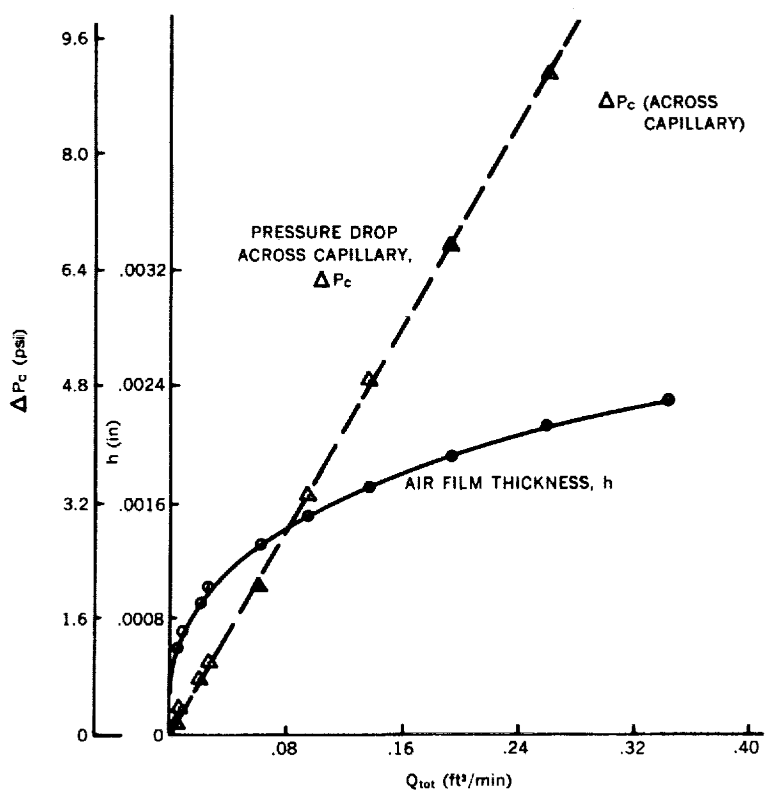

Figure 2—Pressure drop ΔP_c across capillary and air film thickness h as functions of total flow Q_{tot} for the spherical dual-flow bearing

$$p_1 = 2 \ln \frac{R}{r},$$

$$p_2 = 2 \ln \frac{r}{1/4},$$

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which are plotted in Figure 3.

Coefficient of Friction

The theoretical coefficient of friction of the bearing was calculated by dividing the force F required to rotate the bearing at a given speed by the normal load on the bearing. The force F may be obtained from the basic equation

$$F = \mu A \frac{\bar{v}}{h}$$

where A is the area of "contact" and $\bar{v}$ is the average linear velocity of points on the

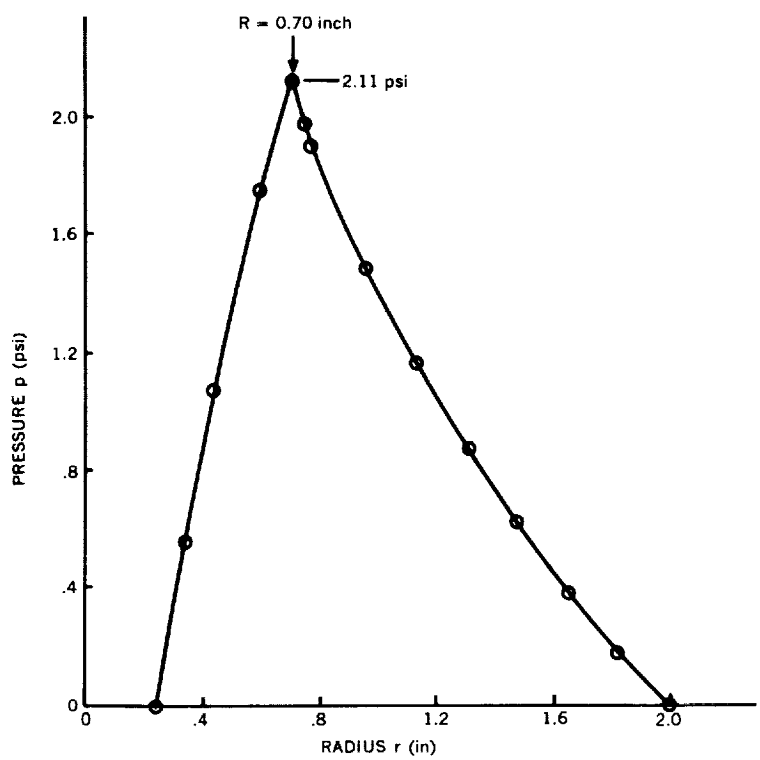

Figure 3 — Radial distribution of pressure within the spherical dual-flow bearing

rotating sphere. For the bearing under consideration $A = 3.32 \text{ in}^2$, $\bar{v} = 7.06$ in/sec (at 60 rpm), and μ and $h = h_0$ have the values given earlier; thus

$$F = 6.1 \times 10^{-5} \text{ lb.}$$

Then, with the normal force $n = W = 15$ pounds, the coefficient of friction is

$$f = \frac{F}{n} = 0.00000406.$$

This shows that, if external drag is neglected, the bearing very closely approaches a frictionless test facility.

Fabrication and Testing

The bearing was fabricated of brass. After being turned on a lathe, the two surfaces were lapped to remove all irregularities and form a smooth surface. The completed bearing was then mounted in a stand, with the 15-pound load suspended from the center, for performance testing at various pressures. Two methods were used to measure the various parameters; one employed a mechanical dial indicator and the other was an electrical method employing a capacitance bridge.

In the first method the bearing and dial indicator were mounted on a stable base. With no air flow through the bearing and a 15-pound weight attached to the bearing, the indicator was adjusted to read zero with its arm on top of the bearing. Air was then introduced at the required input pressure, and the distance the bearing lifted off its seat was noted on the indicator. This measurement was repeated several times. Under the design conditions ($\Delta P_{tot} \approx 4.5$ psi) the indicator measured an average of 0.0008 inch.

The electrical check method employed a capacitance bridge circuit in which the bearing itself was considered a capacitor with air as the dielectric. As the inlet air pressure was changed, the thickness h of the air film varied, changing the value of the capacitance. The air film thickness was calculated from the measured capacitance for several values of input pressure. In Figure 4 the actual film thickness h and ΔP are plotted against the "actual" total flow (calculated from the other measured values but not measured with a flowmeter). For $\Delta P_{tot} \approx 4.5$ psi the film thickness was 0.0007 inch.

Figure 5 shows the physical configuration of the air bearing including the air inlets in which the control capillaries are located.

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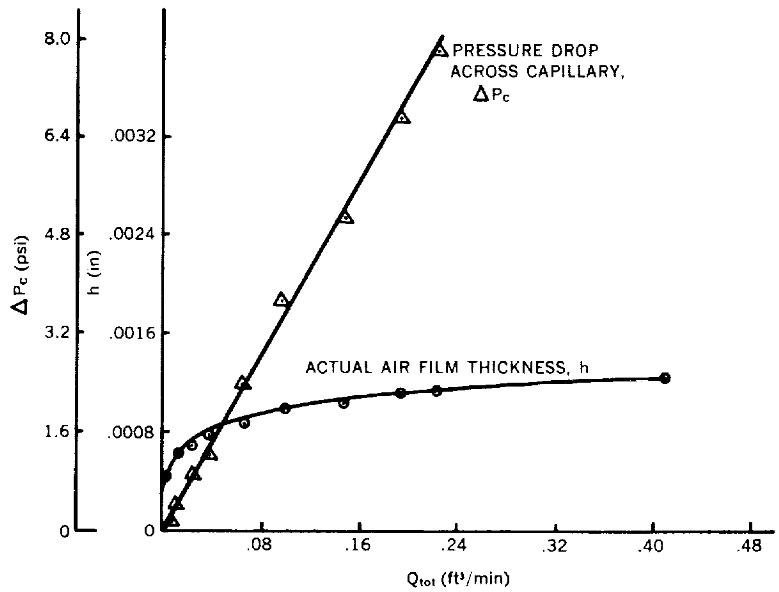

Figure 4 — Actual (measured) air film thickness h and pressure drop across the capillaries as functions of total flow Q_{tot} for the spherical dual-flow bearing

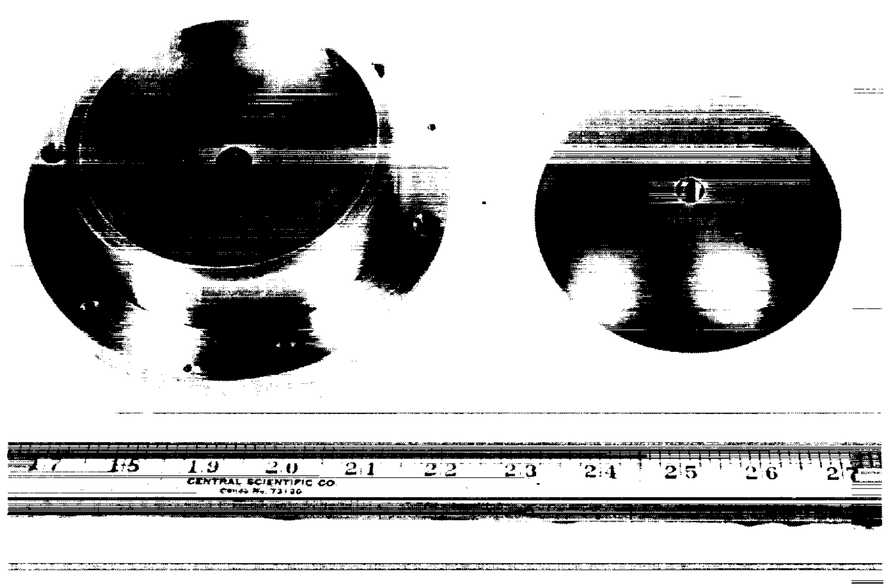

Figure 5 — The spherical dual-flow bearing: socket (left) showing air inlets (small holes) in which the capillary orifices are located; and sphere (right) with load-carrying shaft removed.

CYLINDRICAL DUAL-FLOW AIR BEARINGS

In the process of using the spherical dual-flow air bearing external torques were applied to a test fixture to study the characteristics of the bearing with a rotating body on it. If the torques and center of gravity of the test fixture were not exactly at the center of rotation of the bearing, precession of the rotating sphere would develop. For these particular applications, a cylindrical dual-flow bearing was built on the same principle as the spherical one. However, when the spherical bearing was properly utilized various impulse and momentum characteristics for rotating bodies could be observed and measured.

One of the main advantages for using the dual flow air bearing is that the center of gravity of the system is a considerable distance below the center of curvature of the bearing, providing greater stability.

SPHERICAL MONO-FLOW AIR BEARING

The basic configuration of a spherical mono-flow air bearing can be designed on the basis of a single input capillary opening into an air pocket for maximum supporting capacity, with the outer circumferential rim of the bearing providing the flow control. The cross section and pressure distribution of such a bearing are shown in Figure 6.

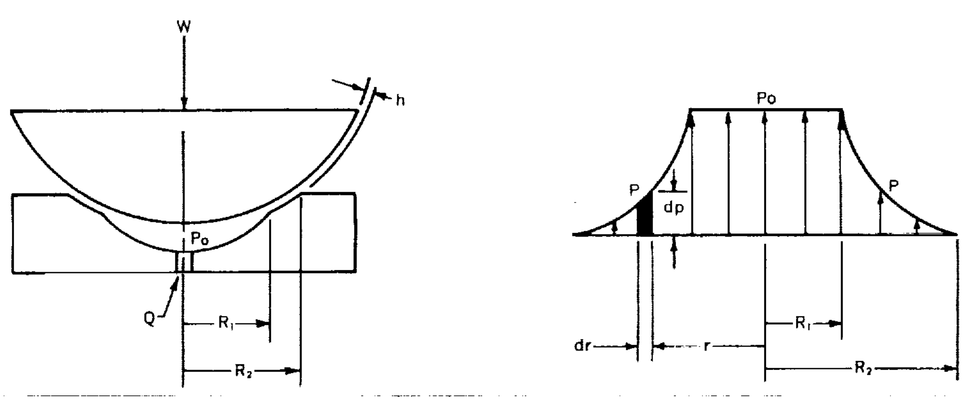

Figure 6 — Configuration and pressure distribution of the spherical mono-flow bearing

This bearing, developed for use in testing the control system of an artificial satellite, was required to support a load of 700 pounds. The large size, weight, and angle of swing made the use of a dual-flow bearing impractical.

Determination of the Pressure Distribution

Since the use of high pressures would complicate the calculations by introducing compressibility effects, the simplest approach to design in this case was to assume a low inlet pressure and then calculate the various radii, rather than the converse.

An examination of Figure 6 with Figure 1 will show that the flow of air is again given by Equation 1 and the element of pressure along the slot by Equation 4, the nomenclature being the same. Thus, the pressure at any point in the slot is

$$p = -k \int \frac{dr}{r}$$

$$= -k \ln r + C \, .$$

To evaluate C, we impose the outer boundary conditions $p = 0$, $r = R_2$:

$$0 = -k \ln R_2 + C \, ,$$

$$C = k \ln R_2 \, .$$

Thus

$$p = k \ln \frac{R_2}{r} \tag{13}$$

where, as in the earlier case,

$$k = \frac{6Q\mu}{\pi h^3} \, .$$

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Total Flow and Peak Pressure

Now an expression for Q may be obtained from Equation 13 by imposing the inner boundary conditions $r = R_1$, $p = P_0$; this results in

$$Q = \frac{P_0 \pi h^3}{6\mu \ln \frac{R_2}{R_1}} . \tag{14}$$

Next the value of P_0 required to support the load W may be obtained from the expression

$$W = P_0 \pi R_1^2 + \int_{R_1}^{R_2} p \, dA$$

where $dA = 2\pi r \, dr$ is the elemental flat projection of the flow-controlling rim area. Substituting from Equation 13 and 14, integrating, and solving for P_0 gives

$$P_0 = \frac{2W}{\pi} \left[\frac{\ln \frac{R_1}{R_2}}{\left(R_2^2 - R_1^2\right)} \right] . \tag{15}$$

This value of P_0 may now be substituted into Equation 14 to give the flow of air required. Since the curvatures of the sphere and socket are equal, the film thickness h varies with r; the average value of h was found to be 0.000673 inch. This results in

$$Q = 169 \text{ in}^3/\text{min}.$$

Determination of the Bearing Radii

Finally, to determine the bearing size, the values $P_0 = 5$ psi, $W = 700$ pounds, and $R_2 = 7$ inches, are chosen and substituted into Equation 15; the result is

$$R_1 = 6.27 \text{ inches}.$$

The design of the single capillary tube is obtained from Equation 12 with $N = 1$. First solve the equation for ℓ :

$$\ell_c = \frac{\Delta P_c \, \pi R_c^4}{8\mu Q} .$$

Let $\ell_c = 30\ R_c$, $\Delta P_c = 4$ psi, and $Q = 2.81$ in^3/sec. Then $R_c = .00519$ inches, and $\ell_c = .156$ inches.

Fabrication and Testing

This bearing was constructed of aluminum and the bearing surfaces were given a Sanford anodic coating for hardness and for increased resistance to the scratching which may result from abrasives or improper handling. The two surfaces were lapped to mirror smoothness. It is most important that the bearing be operated in a clean area free of grit and contamination. Before use, the mating surfaces must be cleaned with a solvent which leaves no residue.

Figures 7 and 8 are photographs of the bearing showing the orifice and the mating surfaces.

The only operational test made on this bearing consisted of plotting the speed and torque vs. time for a given total bearing inertia (Figure 9).

Figure 7—Mating surfaces of the spherical mono-flow bearing

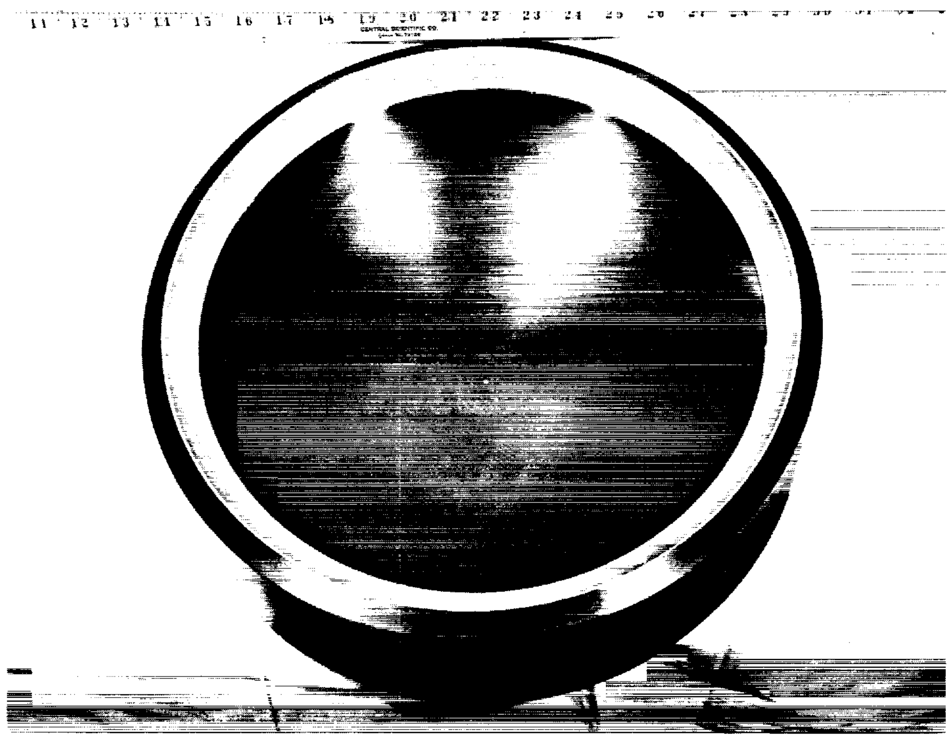

Figure 8 — Interior of the spherical mono-flow bearing socket

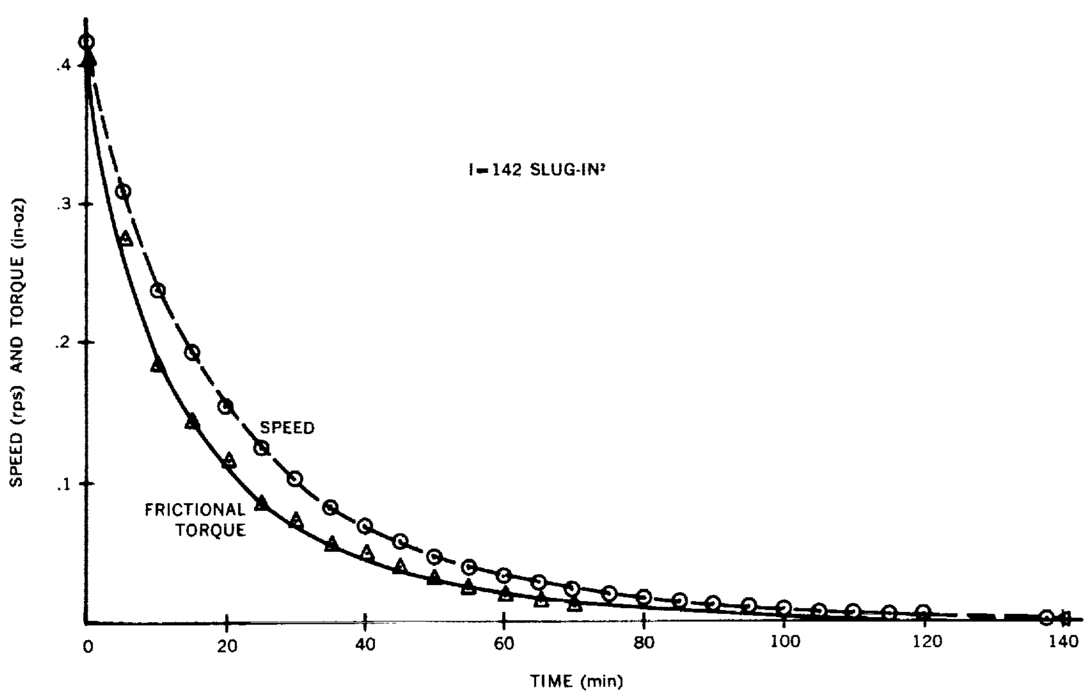

Figure 9 — Coasting characteristics of the spherical mono-flow bearing

FLAT PLATE MONO-FLOW AIR BEARING

A flat plate mono-flow air bearing (Figures 10 and 11) was designed on the same basis as the spherical mono-flow bearing except that the load was assumed to be 400 pounds. This type of bearing has one-plane rotational motion; however, it is very important that the center of mass of the test object be located at the center of rotation of the bearing, and also that the base supporting structure be perfectly level.

Other modifications can be made to this bearing. One is to add sidewalls with orifices, making a cylindrical bearing and preventing any sliding motion due to uneven torque application or an uneven base structure. Another modification would be to utilize the existing air flow to center the bearing by redirecting the exhaust air to act on the side wall of the bearing.

The procedures for hardening and lapping the surfaces were the same as for the spherical mono-flow bearing.

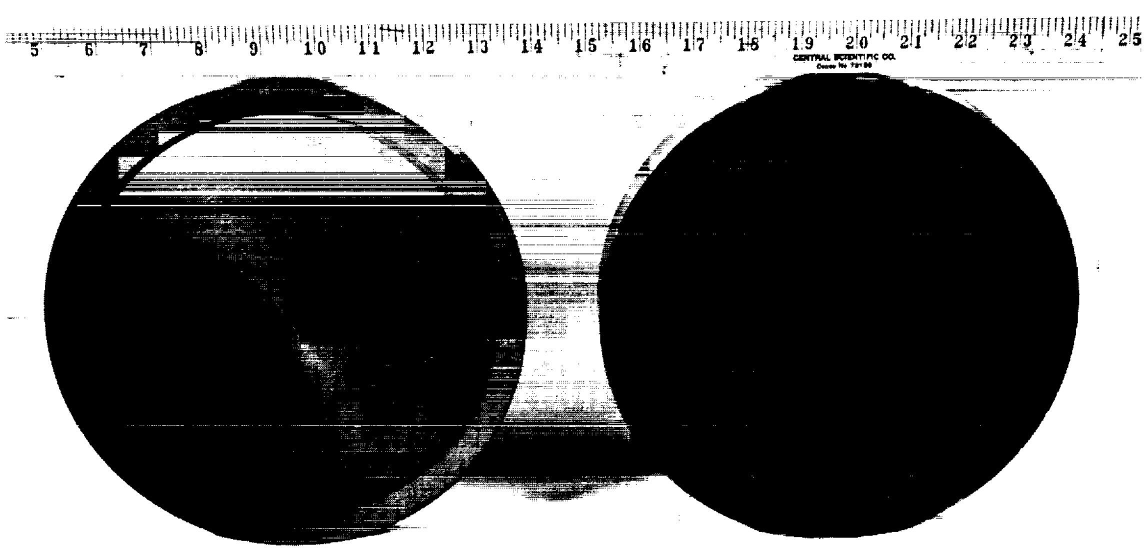

Figure 10—The two components of the flat plate mono-flow bearing

Figure 11 — The flat plate mono-flow bearing, assembled

OPERATIONAL CONSIDERATIONS

Many operational problems are eliminated when a dry, filtered air supply is used. This prevents such contaminants as water, oil, and grit from collecting in the air gap and degrading performance.

Care must be taken, when there is a requirement for frequent intermittent use of the bearing, to provide means of maintaining a space between sphere and socket when the air is shut off. This will prevent accidental damaging of the bearing surfaces.

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